# Rehabilitation of Secondary Heating and Cooling Systems – Case Study

Hui Chen
*Research Associate*
Energy Systems Lab

Song Deng
*Asst. Research Engineer/Asst. Director*
Energy Systems Lab

Homer Bruner Jr. CEM
*Mechanical Systems Specialist*
Energy Office, Physical Plant Dept.

Joel Hugghins
*Research Associate 1*
Energy Systems Lab

Tim Brundige
*Research Technician*
Energy Systems Lab

Dr. W. D. Turner
*Professor/Director*
Energy Systems Lab

Dr. David Claridge
*Professor/Associate Director*
Energy Systems Lab

Energy Systems Laboratory, Texas A&M University System

**ABSTRACT**
Continuous Commissioning (CC[SM]) was performed on the Texas A&M Large Animal Hospital in October 1996 and as a result, significant savings were achieved. Subsequently, the building chilled water and hot water energy consumption increased, and the occupants complained about discomfort problems due to later building operational changes. Most of these problems were caused by control systems in individual areas that were improperly maintained so that coils or other components were operating wildly. When optimal building operations are changed or degrade over time, follow-up CC measures must adapt the systems to maintain efficiency. This degradation may be due to changes in building use, control programming changes, component/sensor failure, or system controls by-pass or override. This paper intends to present the circumstances surrounding this investigation and the subsequent measures taken to correct the problems.

**INTRODUCTION**
The general purpose of Continuous Commissioning is to optimize a building's HVAC system and reduce energy consumption, while simultaneously not compromising occupant comfort. Continuous Commissioning achieves this by modifying existing energy management control systems (EMCS) based on optimal operation schedules and also by repairs of faulty HVAC components and equipment. The success of Continuous Commissioning, however, depends on the persistence of the CC measures and CC savings over many years (Claridge et al., 2002a; Claridge et al., 2002b; Turner et al., 2001). If savings deterioration is detected, attention must return to a building and its operation. A building and its components will degrade naturally over time, but their deterioration can by greatly accelerated by many factors (Chen et al., 2002a, Deng et al., 1998; Deng et al., 2000; Deng et al., 2001). Global building problems, such as pumping issues, cause building-wide problems and therefore tend to cause a more rapid savings decline as well as have the potential for damaging components. In the case of the Texas A&M Large Animal Hospital, significant inroads were initially made in achieving savings on the air-handing unit (AHU) level, but they did not appear to be sustainable. Further investigation uncovered rudimentary issues related to the primary loop and its violent fluctuations in pressure. The building's original system was incapable of insulating its secondary loop from these wild pressures, to the extent that it could not even control secondary $\Delta P$. This violent energy was then distributed throughout the heating and cooling coils of terminal reheats and AHUs. As a result, valves were being damaged and changing conditions made CC measures ineffectual.

Understanding control valve performance is essential in order to: operate the system efficiently, properly tune temperature and pressure sensors, minimize interaction effects on other components of the HVAC system, and to keep the occupants (both humans and animals) comfortable. Several of the characteristics of the valve in operation and criteria for proper valve selection are discussed in several of this paper's references (Avery 1993; Chen et al., 2002b; Fred 1998; Karalus, 1997; Hegberg M. C. 2000; Hegberg R. A. 1997; Rishel 1988).

This paper presents the verification and follow-up efforts, which identified control problems (valve and programming) in secondary pumping systems, and provided recommendations currently being implemented to restore secondary water loop optimization for the Texas A&M Large Animal Hospital complex.

**FACILITY INFORMATION**

The Large Animal Hospital, also known as the Large Animal Clinic, is part of the Texas A&M University School of Veterinary Medicine. It is located behind the Veterinary Research Tower on the north side of west campus in College Station, Texas. This 140,865 square-foot building has two stories and includes a detached isolation ward building and separate breeding center. The Large Animal Hospital is a large medical complex with multi-functional medical facilities. The hospital primarily consists of an administration area, central sterile supply, equine medicine ward, equine surgery, equine exam rooms, equine surgery ward, food animal complex, intensive care unit, pharmacy, radiology section, and veterinary classrooms.

*Figure 1: Front of Large Animal Hospital*

There are seven (7) single-duct, variable air volume (SDVAV) air handling units (AHU) with variable frequency drive (VFD), and four (4) constant air volume (CAV) 100% outside air units. Each main SDVAV AHU has a pre-heat coil, a cooling coil, one supply air fan (total 266 hp), and a return air fan (total 77.5 hp). These seven VAV AHUs provide 80% of the conditioned air area for the building. Each of the 4 CAV 100% outside air units serve the animal wards and has a pre-heat and a cooling coil (total supply fan horsepower: 40hp). There are also 10 fan coil units (FCU) in the electric rooms, mechanical penthouses, and several other locations.

All units except one have heat recovery and preheat coils. Nearly 85% of the terminal VAV boxes use hot water reheat coils and supply air dampers, which are pneumatically controlled.

A schematic diagram of the chilled water system in the building is shown in Figure 2. A schematic diagram of the heating water system in the building is also shown in Figure 3. The campus plant provides chilled water and heating water to the building. In addition there are two (2) parallel chilled water pumps (2×50 hp) and two (2) parallel heating water pumps (2×25 hp). The other two parallel heating water pumps supply a relatively low, constant temperature, supply water to reheat coils in interior zones of buildings. All the pumps in this hospital are with VFD control.

The air handling units, chilled water pumps and FCUs are controlled by a direct digital control (DDC) system.

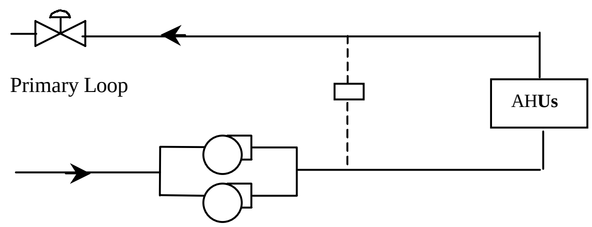



*Figure 2: Schematic diagram of building chilled water system*

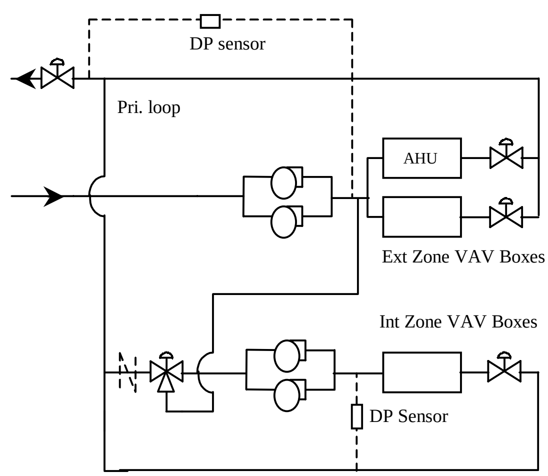



*Figure 3: Schematic diagram of building heating water system*

## INVESTIGATION AND FINDINGS

***Control Valves (AHUs and Pumping Systems)***

The single most important element in any fluid handling system is the control valve. This is because it regulates the flow of fluid to the process. In order to properly select and operate the valve one must understand the design limitations, construction, and geometry of the pressure-absorbing device, and have a firm grasp on the system parameters.

While theoretically, the amount of this pressure drop can be very large, for practical noise, erosion, and wear issues, there must be reasonable limitations. It is necessary to figure out what are the reasons for getting so many bad control valves in less than 10 years building.

***Identified Problems***

It was observed that several AHUs could not maintain the desired cold deck temperatures and operated at temperatures below the cold deck setpoint even with the chilled water valve visibly in the fully closed position. Several AHU control valves, included the main building control valves for hot water and chiller water systems, were either stuck, remaining in the fully open position, or were incapable of modulating.

Nearly 50% of all the control valves at the AHUs were found to be bad. All of the main building control valve heads for chilled water and hot water supply were found to be unoperational as well.

Table 1 shows the malfunctioned control valves and the replaced control valves. This building is only about ten (10) years old. It is a highly unusual thing that nearly 50% of total control valves (AHUs) were replaced during such a short timeframe.

***Life Duration of Control Valve***

Normally valves and valve actuators are designed and tested to achieve a minimum of 100,000 full strokes and over 1,000,000,000 repositions. The number of years of useful life to which this translates is dependent on the HVAC system in which the valve is installed and how frequently the actuator is required to stroke the valve. It is not uncommon for valves, installed in finely tuned HVAC systems, to last more than twenty or thirty years after installation.

***Valve Operational Conditions or Limitations***

There are generally three pressure ratings for the common type of control valves used in building HVAC systems: valve body rating, a close-off pressure rating, and dynamic or modulating rating.

- ***Valve Body Rating***

Valve body rating is the body static pressure rating. The different manufacturers have different values. The valve body ratings under ANSI class 125 are generally less than 200 psi for the chilled water valve and 190 psi for hot water valve (hot water supply temperature is typically less than 200 F). All the control valves in the Large Animal Hospital belong to the ANSI class 125 category.

- ***Close-off Rating***

A valve's close-off rating is based on the power of the actuator. The close-off pressure is frequently many times higher than the dynamic or modulating rating. The close-off pressure value is proportional to the size of actuator and inverse to valve size. The close-off pressure for electronic actuator is 3 ~ 5 times the values for pneumatic actuator. The pressure value of normally opened (NO) valves for pneumatic actuators is less than the value of normally closed (NC) valves, but reverses for electronic actuators. For example, the maximum close-off pressures for normally open valves are 31 psi and 36 psi for the normally closed valves when valve size range is from 2½ inches to 6 inches for pneumatic actuators. Five (5) of the replaced control valves for the main AHUs are 3-inch valves with 8-inch pneumatic actuators. The close-off pressures of the 3-inch valve and 4-inch valve are respectively 29.2 psi and 22.6 psi for the normally open valves under a 18 psi pneumatic control signal (maximum compressed pneumatic pressure to the valve in this building was 18 psi).

- ***Dynamic or Modulating Rating***

Dynamic or modulating rating is the maximum recommended differential pressure for effective modulation. The maximum differential pressure rating is for normal seat and disc wear. Different valve manufacturers have different dynamic ratings, but the maximum recommended differential pressure for modulating is typically 25 ~ 30 psi for valve sizes ranging from 1½ ~ 6 inches with bronze trim. Steel trim extends the modulating pressure range of valves. The maximum recommended differential pressure for modulating in the Large Animal Hospital is 25 psi, because all the valves have bronze trim according to the valve manufacturer information.

***Analysis for Inoperative Control Valves***

It was found that the control valves in this building had been exposed to a very high and frequently fluctuating building differential pressure. The differential pressure was oscillating not only due to lack of building water control, but also from surges in the primary loop on campus of Texas A&M University. The key issue is wild primary differential pressure coupled with bad building control.

- ***Failed existing building water control***

The existing hydraulic pumping systems (hot and chilled water) are two-way variable pumping systems designed to maintain a secondary loop differential pressure. This is the preferred energy savings system, but it must operate as designed. The VFD should operate according to an optimized building differential pressure reset schedule. The optimized building differential pressure reset schedule can be based on the temperature of the outside air since consumption can be linked to outside air temperature. If other, more direct, feedback can be attained from the actual system, a more sophisticated schedule could be developed.

Figure 4 illustrates the building differential pressure fluctuations, chilled water set points, and valve positions in the Large Animal Hospital from February 15, 2001 to February 16, 2001. These drastic fluctuations create intense valve and pumping speed control problems. One can observe that the primary loop differential pressure is usually high, spiking to almost 40 psi and fluctuated all year long. The figure 4 also shows that the main building control valve was constantly hunting and could not modulate well under fluctuating high primary loop differential pressure.

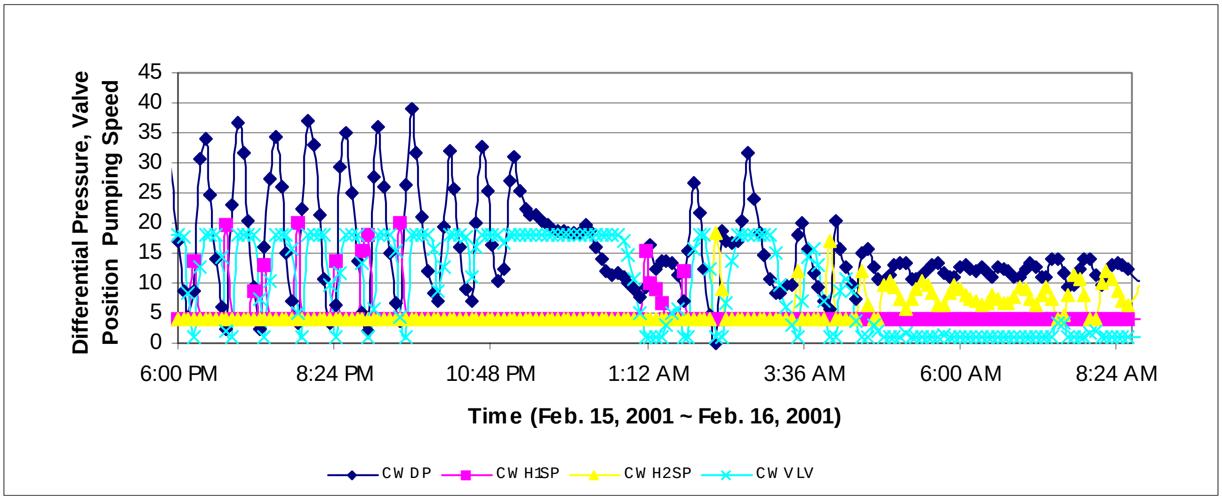


*Figure 4: Building differential pressure, pumping speed and valve position in the Large Animal Hospital during February 15, 2001 to February 16, 2001*

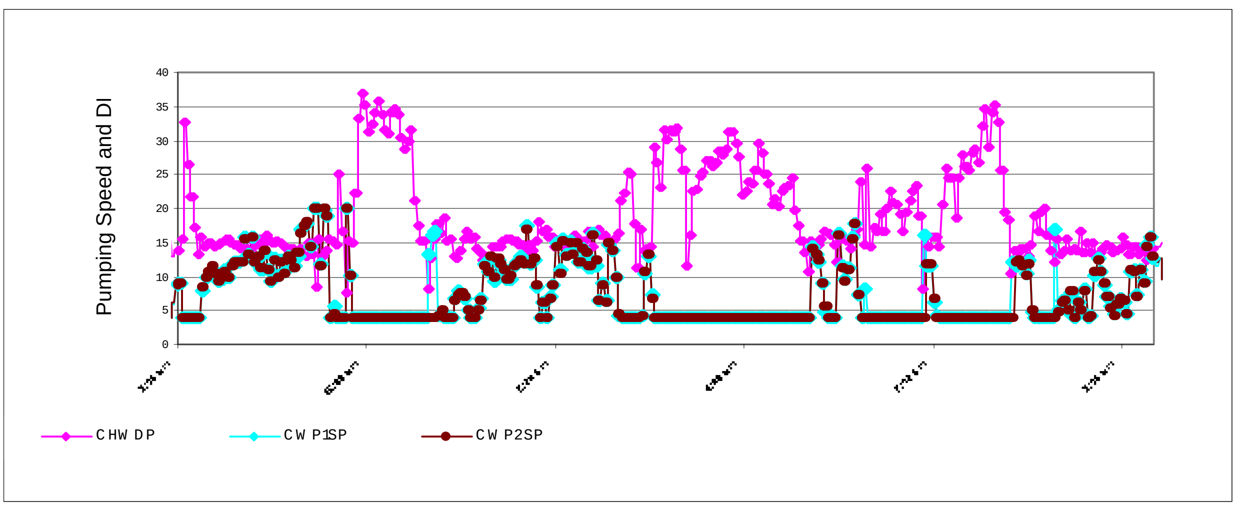


*Figure 5: Primary loop differential pressure and pumping speed control in the Veterinary Research Tower during February 13, 2001 to February 16, 2001*

- ***Fluctuated primary loop differential pressure***

The Texas A&M University campus has encountered numerous problems related to thermal distribution (Deng et al., 2001). The campus has long had difficulty with thermal distribution to buildings served by the central plants. Pressure problems were common in the chilled water and hot water distribution system during peak demand periods. The differential pressure between supply and return headers at buildings far from the central plants was frequently negative, in the middle area was neutral, and close to the plant was positive – causing poor circulation to buildings along A&M's extremities.

Figure 5 shows how the primary loop differential pressure fluctuated nearby the Large Animal Hospital during February 13, 2001 to February 16, 2001. The trended differential pressure data were from the Veterinary Research Tower, which is located in front of the Large Animal Hospital on the north side of west campus. When the main building control valve was commanded to the fully open position, building pumps were used to maintain chilled water differential pressure at a set point of 14 psi. Differential pressures higher that 14 psi (such as those caused by the oscillations of the primary loop) caused the pumps to stop. When the differential pressure from the primary loop was lower than set point, the pumps were commanded to run again. The figure shows that the primary loop differential pressure was oscillating dramatically and sometimes reached as high as 37 psi.

- ***Valve pressure drop ($\Delta P_v$) and system pressure drop ($\Delta P_s$)***

Assume that the building **m**ain differential pressure ($\Delta P_m$) is always constant using variable speed pumping. The building main differential pressure ($\Delta P_m$) must then equal the combination of the **s**ystem pressure drop ($\Delta P_s$) and the fluid control **v**alve pressure drop ($\Delta P_v$).

$$\Delta P_m = \Delta P_s + \Delta P_v \qquad (1)$$

The coil and its associated piping (except control valves) constitute a relatively small system (see Figure 6: schematic diagram of a cooling/heating coil). The pressure drop through the system then varies as the square of the flow according to the law of hydraulics:

$$\Delta P_1/\Delta P_2 = (Q_1/Q_2)^2 \qquad (2)$$

It useful then to note that control valve flow capacities are expressed in terms of the pressure drop across the valve and the flow coefficient, $C_v$, which is defined as the flow in gpm (Q) through the wide-open valve with a 1 psi pressure drop. The valve flow coefficient may be determined from the formula:

$$Q = C_v\,(\Delta P)^{1/2} \qquad (3)$$

Where Q is the coil's design flow rate in gpm and $\Delta P$ represents the pressure drop across only the control valve in psi.

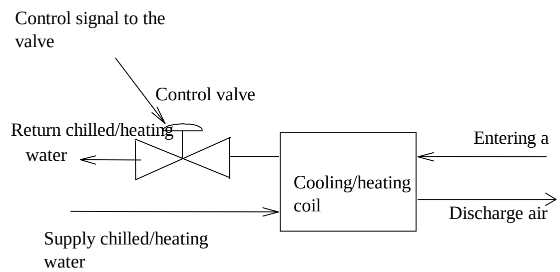


*Figure 6: Schematic diagram of a cooling/heating coil*

Based on above theoretical formulas, calculation and simulation were performed for valve differential pressure ($\Delta P_v$) and system pressure drop ($\Delta P_s$) at the design flow rate and over the design flow rate. An existing 4-inch valve ($C_v$: 160) with an 8-inch pneumatic actuator (from AHU HP-6) was used for a sample calculation and simulation. The ratio of $\Delta P_v$ to $\Delta P_s$ in this case is assumed to be 0.5 and $\Delta P_s$ to be 12 psi when at the design flow rate. For the simulation, $\Delta P_v$ will be 6 psi and $\Delta P_m$ will be 18 psi. Recall that $\Delta P_m$ is assumed to be constant. The design fluid flow rate for the system is 392 gpm, and $C_v$ for the control valve will be 160 gpm. If the coil load were to decrease to a point where a flow rate of 352 GPM is needed, the $\Delta P_s$ would also decrease to 9.7 psi. Then $\Delta P_v$ will reach 8.3 psi because $\Delta P_m$ must remain a constant 18 psi. $C_v$ must then be 122.6 gpm. This shows a decrease of 76% in the value of $C_v$ and a decrease of 80% in the value of $\Delta P_s$. Figure 7 illustrates the above $\Delta P_v$ and $\Delta P_s$ at different percentages of design flow.

Field survey and trended data shows that the building constantly experiences immense differential pressures - around 35 psi or higher. A simulation for the same valve under an excessive building differential pressure of 35 psi was performed while keeping the valve information the same as the simulation at a different design flow rate. Figure 8 shows the system (coil and pipe) pressure drop and the valve pressure drop versus flow rate at excessive building pressure $\Delta P_m$. Even though the differential pressure for the coils was not a stable 35 psi, the simulation assumes 35 psi for simplification.

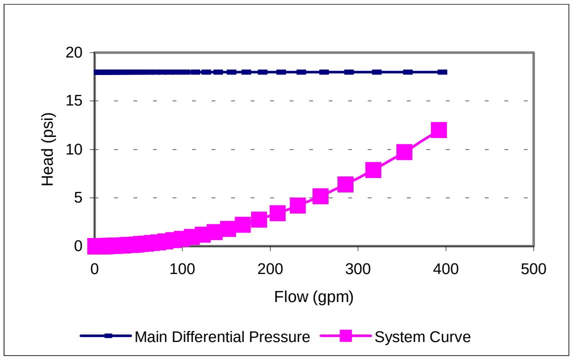


*Figure 7: Valve and coil system performance curve under design conditions (initial design ratio: $\Delta P_v/\Delta P_s = 0.5$)*

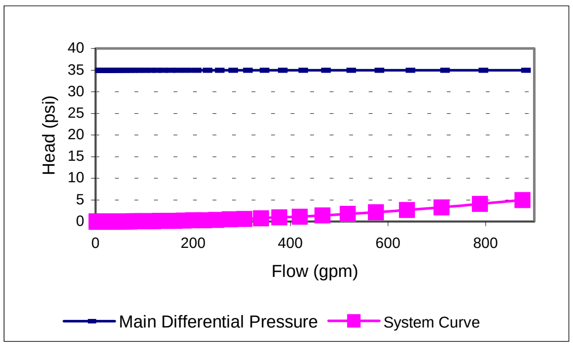


*Figure 8: Valve and coil system performance curve under huge building pressure*

Compare the design simulation result (Figure 7) with the huge building pressure situation (Figure 8). A 4-inch control valve consumes more than 30 psi at different flow rates under the 35 psi of building differential pressure. The control valve only consumes from 6 ~ 18 psi at different flow rates under the original design settings.

- ***Reasons for damaged components***

In order to restrict the flow of liquids and to maintain water differential pressure set points, control valves develop a pressure drop across their ports. Each control valve in the Large Animal Hospital had been subjected to radically fluctuating differential pressures. Analysis on the primary loop and secondary loop shows that the primary loop had a long-term fluctuation problem, but the main building control valves did not restrict the hot and chilled water flow, thus maintaining the water differential pressure set point for the building. This caused the AHU control valves to deal with the unfettered differential pressure and to operate outside of the valves' operation limitations on modulation, erosion, cavitations, wear, and close-off pressure.

The valve would assuredly have a shortened life if it were installed in a system where the available dynamic pressure of the fluid approached that of the valve close-off pressure. Additional wear would occur if the dynamic pressure exceeds the dynamic rating of the valve itself. For example, the 4-inch valve ($C_v$: 160 and NO) with an 8-inch pneumatic actuator (under 18 psi pneumatic control signal) from AHU HP-6, has only 22.6 psi of close-off pressure and 25 psi of dynamic pressure. The close-off pressure is less than the dynamic pressure value. Even worse, actual differential pressure was normally in the range of 30 psi to 40 psi, which was higher than both the maximum recommended dynamic differential pressure value and the close-off pressure limitation. The simulation for valve pressure drop and system pressure drop under large differential pressure shows that control valve has to consume more pressure drop (far higher than dynamic requirement and close-off pressure limitations) than the design condition. The control valve under high $\Delta P_v$ (e.g. higher than 30 psi, see Figure 4) is assumed to be damaged much faster than would occur under normal conditions.

If the controls system forces valves to operate outside of the valve operation limitations in the midst of significant differential pressure fluctuations, damage will occur unless system adjustments are made (Hegberg Mark C. 2000). In extreme cases, damage caused by erosion, cavitations, and a process akin to wire drawing can occur. In a simple system, the actuator might simply stop functioning properly, forcing flow in a pneumatic system (e.g. main building control valve actuators).

Cavitation can occur when the pressure drop across the control valve opening is excessive. The combination of pressure drop through the plug and velocity increase across the orifice cause the pressure on the surface of the liquid to fall below its vapor pressure. Tiny steam bubbles are generated during this process, which are mixed into the turbulent liquid. The steam bubbles then implode with tremendous force as the pressure recovers downstream. The limited area where this occurs can cause significant damage to the valve and plug (ASHRAE, 2000b). The risk of cavitation tends to occur most in hot water systems when taking pressure drops greater than 15 psi across the control valve (Hegberg Mark C. 2000).

This high building differential pressure was causing high flow and resulting in chilled water valves leaking by. The water capacity is proportional to differential pressure. For example, the water flow for a 4-inch valve ($C_v$: 160 and NO) of AHU HP-6, is 392 GPM under 6 psi of differential pressure and will be 876 GPM under the value of 30 psi based on the manufacturer's catalog data.

***CC Measures & Follow-up***

Several CC measures were taken based on above findings and investigations:

- Replaced the non-functional control valves and failed pneumatic valve actuators (see Table 1)
- Installed new electronic actuators (instead of replacing the pneumatic actuators) for the main building control valves and for several control valves on the AHU level. The installation of electronic actuators results in an increase of the close-off pressure because the close-off pressure of electronic actuators is far higher than that of pneumatic actuators.
- Reset differential pressure setting for hot water and chilled water systems
- Tuned PID loop for stable control of secondary water system and tertiary reheat water system.
- Enhanced the economizer controls. The economizer controls were set to operate below 50°F outside temperature to maintain 55°F cold deck temperature. The economizer cycle has been changed to start when the outside air temperature dips below 65°F and maintains cold deck setpoint by modulating the maximum outside dampers and return dampers simultaneously.
- Modified existing control program.
- Rescheduled cold deck statement.

***Reheat Pumping Systems***

***Identified Problems***

The Large Animal Hospital Building was fitted with three-way control valves designed to support tertiary, variable speed, hot water, pumping systems. These systems typically supplied a relatively low (140°F), constant temperature, supply water to reheat coils in interior zones of buildings. The intent of the three-way valves was to allow the formation of a closed loop reheat system whose temperature was a blend of primary loop hot water and reheat water return.

*Table 1 Replaced Control Valves for the Large Animal Hospital*

| Served Unit | Original valve & actuator | Replaced valve & actuator |
|---|---|---|
| HW-2 chilled water | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator |
| HW-3 chilled water | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator |
| HW-3 Pre-heat | 2-way, NO, $C_v$: 63, 2 ½" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 63, 2 ½" line; Electronic actuator |
| HP-1 chilled water | 2-way, NO, $C_v$: 160, 4" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 160, 4" line; 12" pneumatic actuator |
| HP-3 chilled water | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator | -way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator |
| HP-4 chilled water | 2-way, NO, $C_v$: 63, 2 ½" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 63, 2 ½" line; Electronic actuator |
| HP-5 chilled water | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator |
| HP-6 chilled water | 2-way, NO, $C_v$: 160, 4" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 160, 4" line; Electronic actuator |
| HP-7 chilled water | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 100, 3" line; 8" pneumatic actuator |
| Reheat heating water system | 3-way, $C_v$: 100, 3" line; 8" pneumatic actuator | 2-way, NO, $C_v$: 100, 3" line; Electronic actuator |
| Main building chilled water valve | 2-way, NO, 10" line; pneumatic actuator, | 2-way, NO, 10" line; Electronic actuator |
| Main building heating water valve | 2-way, NO, 6" line; pneumatic actuator | 2-way, NO, 6" line; Electronic actuator |

Figure 3 shows the original reheat-pumping diagram. Whenever the control program tried modulate the 3-way valve, water hammer noise would result from the valve operation. To bypass this issue the facility staff opened the supply water port allowing it flow fully through the valve and therefore shut off return water to the valve. The supply water was unrestrained fed directly to the reheat coils. This problem has been an issue since the building was built some 10 years ago. Other such 3-way valve systems do not work either. Several investigations were performed over the years without finding a solution to this problem.

***Analysis for the original reheat system***

- ***Reason for failed 3-way valve***

The 3-way control valve for this reheat system is designed to be a mixing valve. As figure 2 shows, the common line of the 3-way valve to the reheat pumps and the supply water line to the valve is normally open and return water port is normally closed. Supply water (150°F ~ 170°F) supposedly mixes with return water in the valve and then the mixed water (about 140°F) travels through the common line to the coil. The differential pressure between supply water and return water around the 3-way valve is at least 15 psi ~ 18 psi. Whenever the controller sent a signal to the valve to modulate the valve's position for mixing water temperature, the 15 to 18 psi of differential pressure generated the hammer noise. It was found that the design and installation of a check valve was neglected on the secondary return water line to the 3-way control valve. The missing check valve proved detrimental to controlling flow into this tertiary loop.

- ***Comparison of two-way and three-way control valves for tertiary, variable speed***

Should a missing check valve be installed on the return water piping or should a new two-way control valve be installed instead of the original three-way valve? Determining which piping configuration and control scheme is the most effective method of controlling hot water distribution to the terminal reheat coils is the undertaking of this section. The conclusion will be determined from a detailed comparison of two-way and three-way control valves for tertiary, variable speed pumping systems. The following factors were considered to make determination:

- Energy consumption
- Maintenance concerns
- Regulating supply temperature to coil
- Initial implementation cost
- Sustained operational cost

The two-way control valve provides a variable flow and constant differential temperature, while the three-way control valve keeps constant flow and variable differential temperature (ASHRAE, 2000a). For the tertiary variable pump system, the energy consumption of the two-way control valve will be less than the energy consumption of the three-way control valve (even though the difference is only slight). The reheat coils are located in the terminal boxes, not radiators in the room. The heating water supply temperature is just about 170°F in the winter time and 140°F during the summer on campus of the Texas A&M University. This portion of the system is a low-temperature water (LTW) system range. The three-way control valve exists primarily for temperature control (ASHRAE, 2000b). It was found that regulating the supply heating water temperature for reheat coils is not necessary. Therefore, for stability and simplification, the two-way control valve scheme is the preferred device for this reheat system.

***CC Measures & Follow-up***

In one recent piping renovation, the three-way control valve was removed and replaced with a two-way control valve for the variable speed pumping system. Figure 9 shows the repiped layout of the reheat pumping system. The pumping control program was modified based on the renovation.

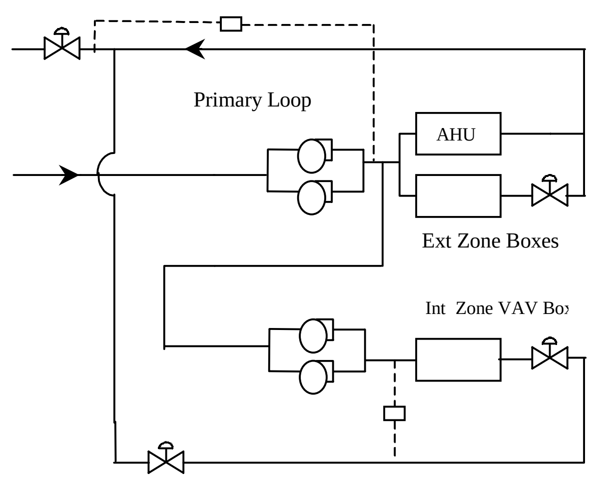


*Figure 9: Schematic diagram of building hot water system after CC follow-up*

***Savings Analysis***

Significant energy savings were achieved through Continuous Commissioning and the CC follow-up performed on the hospital complex. Hourly whole-building chilled water and hot water energy consumption data was retrieved from the energy management database to quantify the verification. Figures 10 – 11 show the energy consumption difference between pre- & post - rehabilitating the secondary heating and cooling systems. It is obvious that the energy consumptions of post – CC follow-up for both chilled water and hot water are less than the values of the pre – CC follow-up. The different data pattern affirms the savings though the retrieved data is over a short period (a month).

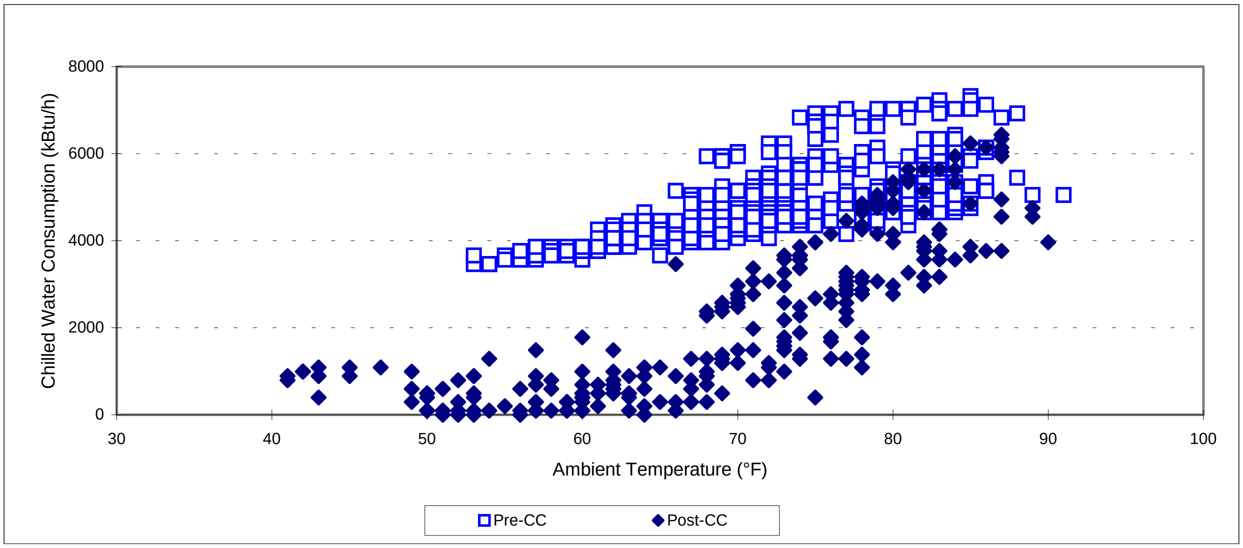


*Figure 10: Chilled water consumption versus ambient temperature for pre- & post - rehabilitating the secondary heating and cooling systems*

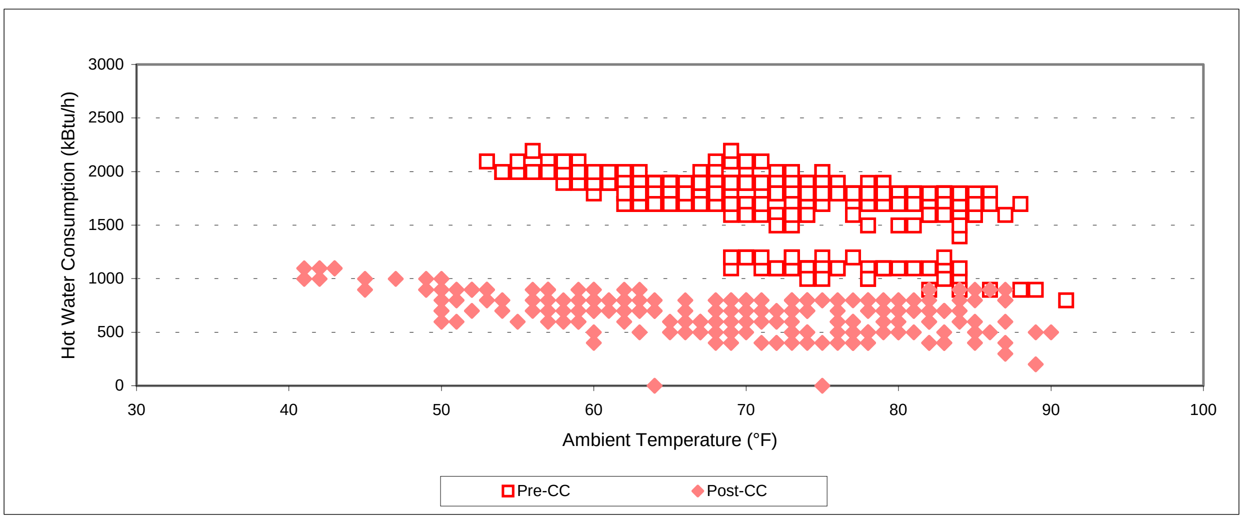


*Figure 11: Heating water consumption versus ambient temperature for pre- & post - rehabilitating the secondary heating and cooling systems*

## CONCLUSIONS

The secondary heating and cooling systems' operation have been improved by CC follow-up. Savings after rehabilitating the secondary heating and cooling systems return to the building, after damaged control valves and actuators were replaced, EMCS programs were modified, and the reheat system was repiped for replacing the failed 3-way valve with the new 2-way valve. This rehabilitation, as a CC follow-up, shows that CC verification and follow-up can sustain persistence of energy savings and optimization for building HVAC operation.

## ACKNOWLEDGEMENTS

The work in this paper was sponsored by Texas A&M University, Physical Plant, Utilities Office of Energy Management. We greatly appreciate the support for this work from the Office of Energy Management and the Continuous Commissioning Group, Energy Systems Laboratory, Texas A&M University.

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